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SIMULATION OF LAMINAR CONVECTION FLOW OF AL_2O_3-WATER NANOFLUID IN AN ASYMMETRIC HEATED CHANNEL

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ABSTRACT The present paper proposes a two dimensional analysis of the laminar convection flow of water-Al_2O_3 nanofluid inside a rectangular section channel with non-symmetric boundary conditions. In particular, a constant heat flux is applied on the top surface of the channel and an adiabatic condition on the bottom one. This situation is typical of many devices, such as solar collector, which receives thermal radiation from the top surface with objective to heat a working fluid, water-Al_2O_3 in the present case, and which are insulated on the bottom one in order to limit energy waste.
Numerical simulations are developed by using the commercial software COMSOL, which employs finite element methods approach to solve the conservation equations, moreover thermal dependent properties are considered in the simulations The analysis is conducted for Re numbers ranging between 500-1000, concentration between 0%-6% and dimension of particles between 20-60 nm.
An increase of Nu is observed at the increase of the concentration, as well as a substantial increment of the pressure losses.

INTRODUCTION

Nanofluids are a new category of fluids consisting in a suspension of metallic nanoparticles, generally metal oxidises, and a liquid, such as water, glycol, etc. The combination of a liquid with metallic nanoparticles allows to enhance the average thermal conductivity of the liquid, thus improving its thermal performances. On the other hand, there is also an increase of the viscosity, therefore a higher pumping power is required to move the fluid.
This idea is not new, in fact it was introduce by Maxwell at the end of 19th century [Bianco et al., 2015], but during that time it was only possible to produce particles with dimensions in the order of micrometers or millimetres, therefore he experiences relevant problems in terms of huge increase of viscosity and abrasion of pipes surfaces. Today the available manufacturing capabilities allow to produce particles with dimensions in the order of nanometers and the problems detected in the first experiments conducted by Maxwell have been not noticed.
Different authors studied forced laminar convection of nanofluids in pipes or channels with uniform temperature or heat flux applied on the walls. The investigations have been developed both experimentally and numerically with the main focus to estimate the heat transfer enhancement due to the utilization of nanofluids.
For example Zae et al. [2016] proposed a numerical analysis of the laminar forced convection of Al_2O_3-water nanofluid within a flat tube. They analysed the impact of nanoparticles on the Nusselt number and evaluated the global performance by developing an entropy generation analysis.
Whereas, Utomo et al. [2014] presented an experimental investigation on the forced laminar convection of Al_2O_3, TiO_2 and carbon nantubes water based nanofluids. They concluded that all the investigated nanofluids follows standard heat transfer correlation developed for liquids within ±10%.

The present paper aims at developing a numerical investigation on the laminar forced convection flow of Al_2O_3-water nanofluid within an asymmetric heated channel. This configuration is typical in hybrid solar panels, where there is a heat flux on the top of the plate, whereas the bottom part is insulated. The investigation is developed for Re equal to 250, 500 and 100, for a concentration of the nanoparticles within 0%-6%, for particles dimensions of 20 nm and 40 nm and for q=1000 W/m^2. Temperature profiles and heat transfer coefficients are determined and discussed for all the proposed cases.

NANOFLUID PROPERTIES

The definition of thermophysical properties of nanofluids represent a key step of the simulation process. As shown by Das et al. [2008], the classical rules of mixture do not apply for nanofluids and specific relations are needed, especially for thermal conductivity and dynamic viscosity. In the present work, the following equations are utilized to evaluate Al_2O_3-water nanofluid thermophysical properties:

$$\rho_m = (1-\varphi)\rho_{bf} + \varphi\rho_p \tag{1}$$

$$\left(C_p\rho\right)_m = (1-\varphi)\left(C_p\rho\right)_{bf} + \varphi\left(C_p\rho\right)_p \tag{2}$$

Eq. (1) is calculated according to the general rule of mixtures [Das et al., 2008], whereas Eq. (2) is calculated by assuming thermal equilibrium between the nanoparticles and the base fluid [Khanafer and Vafai, 2011].
Viscosity and thermal conductivity are evaluated on the basis of the correlations proposed by Corcione [2011, 2012]. They are based on the analysis of a large amount of experimental data available in the literature:

$$\frac{\lambda_m}{\lambda_{bf}} = 1 + 4.4\cdot \mathrm{Re}_d^{0.4}\cdot \mathrm{Pr}^{0.66}\cdot\left(\frac{T}{T_{fr}}\right)^{10}\cdot\left(\frac{k_p}{k_{bf}}\right)^{0.03}\cdot\varphi^{0.66} \tag{3}$$

In Eq. (3) Pr is the Prandtl number of the base liquid and Re_d is the nanoparticle Reynolds number [Corcione, 2012]:

$$\mathrm{Re}_d = \frac{\rho_{bf}\cdot v_b\cdot d}{\mu_{bf}} \tag{4}$$

Where v_b is the Brownian velocity, defined as:

$$v_b = \frac{2\cdot C_B\cdot T}{\pi\cdot\mu_{bf}\cdot {d_p}^2} \tag{5}$$

Instead, the dynamic viscosity is defined as Corcione [2011]:

$$\frac{\mu_m}{\mu_{bf}} = \frac{1}{1-24.3745\cdot d^{-0.264}\cdot\varphi^{1.028}} \tag{6}$$

It is important to note that Eqs. (3) and (6) account for both concentration and diameter of nanoparticles.

NUMERICAL MODEL

In the present paper a numerical simulation of developing laminar convection flow of Al_2O_3-water nanofluid within an asymmetrically heated channel is proposed. The nanofluid is assumed as a homogenous liquid with modified properties, as given in equations (1-3,6), therefore mass, momentum and energy equations assume the following form:

$$\overline{\nabla} \cdot \overline{V} = 0 \tag{7}$$

$$\rho_m \left(\overline{\nabla} \cdot \overline{V}\right)\overline{V} = -\overline{\nabla} p + \mu_m \nabla^2 \overline{V} \tag{8}$$

$$\rho_m Cp_m \left(\overline{V} \cdot \overline{\nabla}\right)T = k_m \nabla^2 T \tag{9}$$

The considered geometry is reported in Figure 1 and the domain is assumed two dimensional, with thickness of 0.01 cm and a length of 2 m. A constant heat flux is imposed on the top surface, whereas the bottom is adiabatic. The inlet temperature is fixed at 293 K and the inlet velocity is calculated in various conditions in order to have Re number equal to 250, 500 and 1000 in all the considered cases.

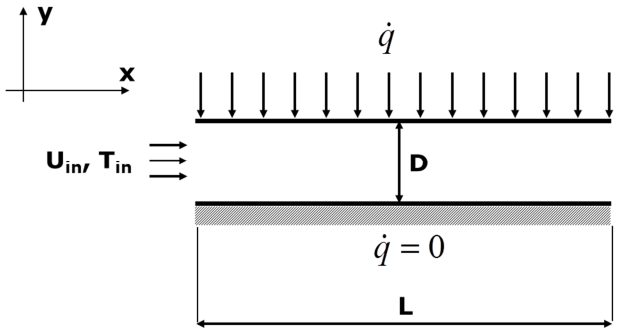

Figure 1. Schematic of the configuration under investigation.

The problem is solved by employing the finite elements method by using the software COMSOL Multiphysics. Before proceeding with the calculations, a sensitivity analysis has been developed by comparing three different structured grids with 18x300, 35x600 and 70x1200 nodes respectively and the most convenient one in terms of results accuracy and solution time has been considered. In particular, the optimal grid resulted to be the 35x600, because it minimizes solution time, while guaranteeing a high level of accuracy.

Moreover, the computational grid has been also subjected to validation by comparison with experimental data. To this scope, an inward heat flux of 5000 W/m^2 was applied on the top and bottom surface, in order to compare the local Nusselt number calculated numerically with the Shah's equation, as suggested by Utomo et al. [2014]:

$$Nu_x = \begin{cases} 1.302 \cdot x^{*\frac{-1}{3}} - 1 & x^* \le 0.00005 \\ 1.302 \cdot x^{*\frac{-1}{3}} - 0.5 & 0.00005 \le x^* \le 0.0015 \\ 4.364 + 8.68 \cdot \left(1000 \cdot x^*\right)^{-0.506} & \exp\left(-41 \cdot x^*\right) \cdot x^* > 0.0015 \end{cases} \tag{10}$$

where x^* is equal to:

$$x^* = \frac{x / D_i}{\mathrm{Re} \cdot \mathrm{Pr}} \tag{11}$$

In the present analysis D_i is the hydraulic diameter, calculated by considering a depth of the channel equal to 16 cm.

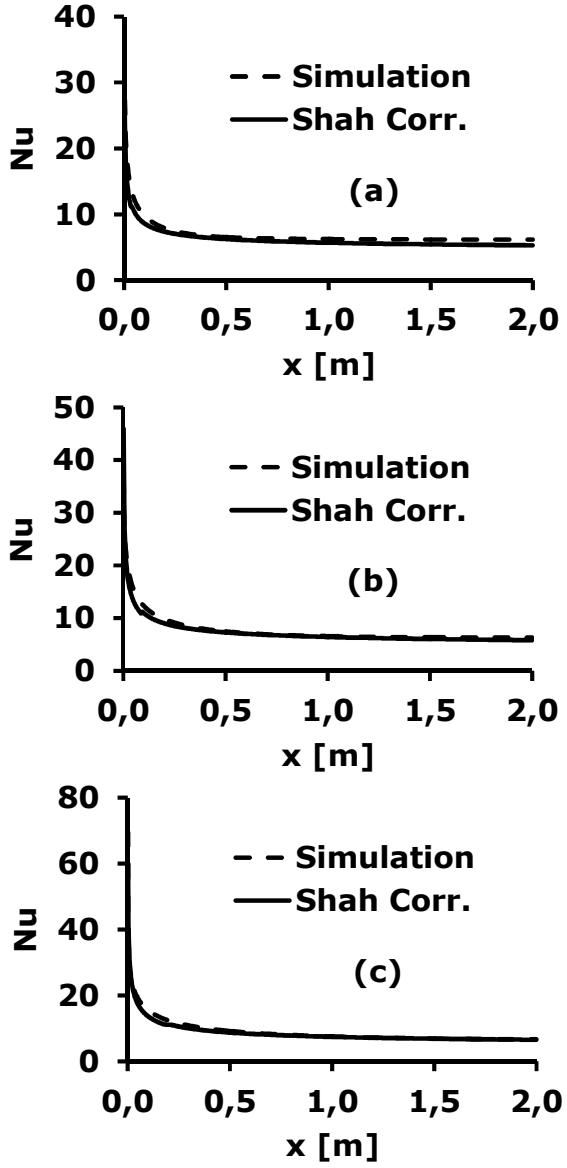

Figure 2. Comparison of the local Nusselt number obtained from the simulation with the experimental correlation proposed by Shah [Utomo et al., 2014] for an inward heat flux of 5000 W/m^2 imposed on both the wall and for different Re: (a) 250; (b) 500; (c) 1000.

By analysing Fig. 2, it is possible to observe that there is a very close correspondence between the numerical simulation and the experimental correlation. In particular, the considered grid has an error on the average Nu always less than 10%, namely 9% for Re=250, 5% for Re=500 and 3% for Re=1000, therefore it can be concluded that the numerical model is accurate and it can be utilized for the analysis of other cases.

RESULTS AND DISCUSSION

Results are reported in terms of temperature, heat transfer coefficient and Nu number profiles for concentration of 2%, 4% and 6%, diameters of 20 and 40 nm, q=1000 W/m^2 and Re numbers of 250, 500 and 1000. For comparison purposes also the results for the base fluid at Re 250, 500 and 1000 are reported.

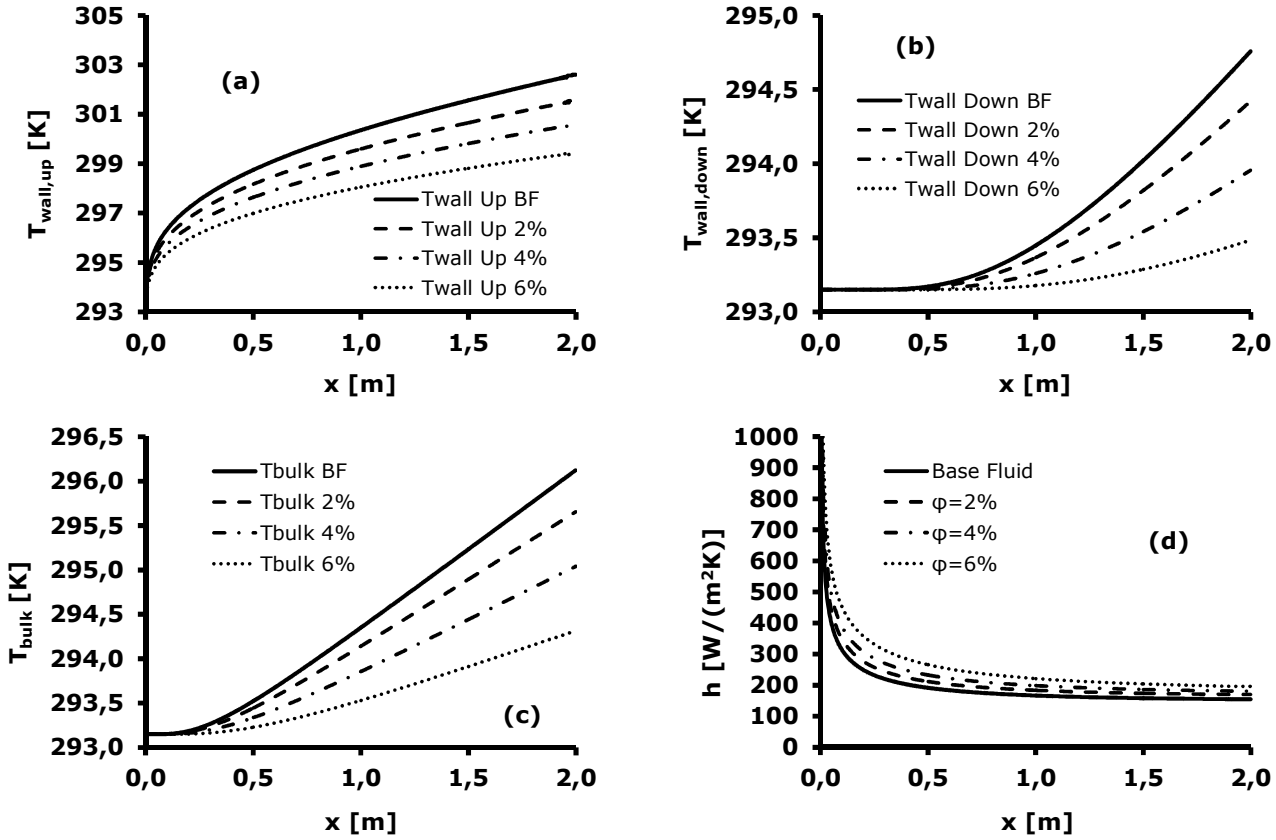

Figure 3. Temperature and heat transfer coefficient distributions for Re=250 and d=20 nm: (a) temperature on the top wall; (b) temperature on the bottom wall; (c) bulk temperature; (d) local heat transfer coefficient

Figure 3 reports temperature distributions and heat transfer coefficient for Re=250 and particles with a diameter of 20 nm. Figure 3(a) shows that at the increase of particles concentration there is a decrease of the top wall temperature. This means that nanofluids allow to cool the surface in a more effective way with respect to the base fluid, i.e. water. A similar trend is also detected for the temperature on the bottom wall, Fig. 3(b), and for the bulk temperature, Fig. (3).

The analysis of Fig. 3(b) highlights that till x=0.5 m, the temperature does not increase in the bottom wall due to the fact that the thermal disturbance in the initial part of the channel is not able to reach the bottom surface. A similar behaviour, but with a reduced impact, is observed also for the bulk temperature in Fig. 3(c), in fact for x<0.15 m the temperature is approximately constant and equal to the inlet condition. This means that the thermal disturbance in the initial part of the channel is confined to the top surface.

Finally, Fig. 3(d) reports the local heat transfer coefficient profiles. It can be noticed that the higher is the concentration and the higher is the value of the coefficient. This means that nanofluids enhance the heat transfer capabilities of the base fluid, water in this case. Therefore, if the objective is to increase the heat transfer rate, it can be said that nanofluids have a beneficial effect.

From the analysis of the figure it can also be concluded that the flow is fully developed, in fact the heat transfer coefficient for x>1.0 m is substantially constant.

The effect of nanoparticles dimension is shown in Figure 4, which compares the results at Re=1000 for base fluid and for nanofluid with particles of diameters 20 and 40 nm.
Fig. 4(a) reports the temperature on the top wall and highlights the fact that particles with lower diameters have a higher cooling effect. This is due to the fact that the lower is the dimension of the particles, the higher is the amount of particles necessary to reach a determined concentration, therefore the surface of interaction of smaller particles is higher than that of larger particles. This enhanced surface of interaction between particles and base fluid determines a stronger cooling effect.

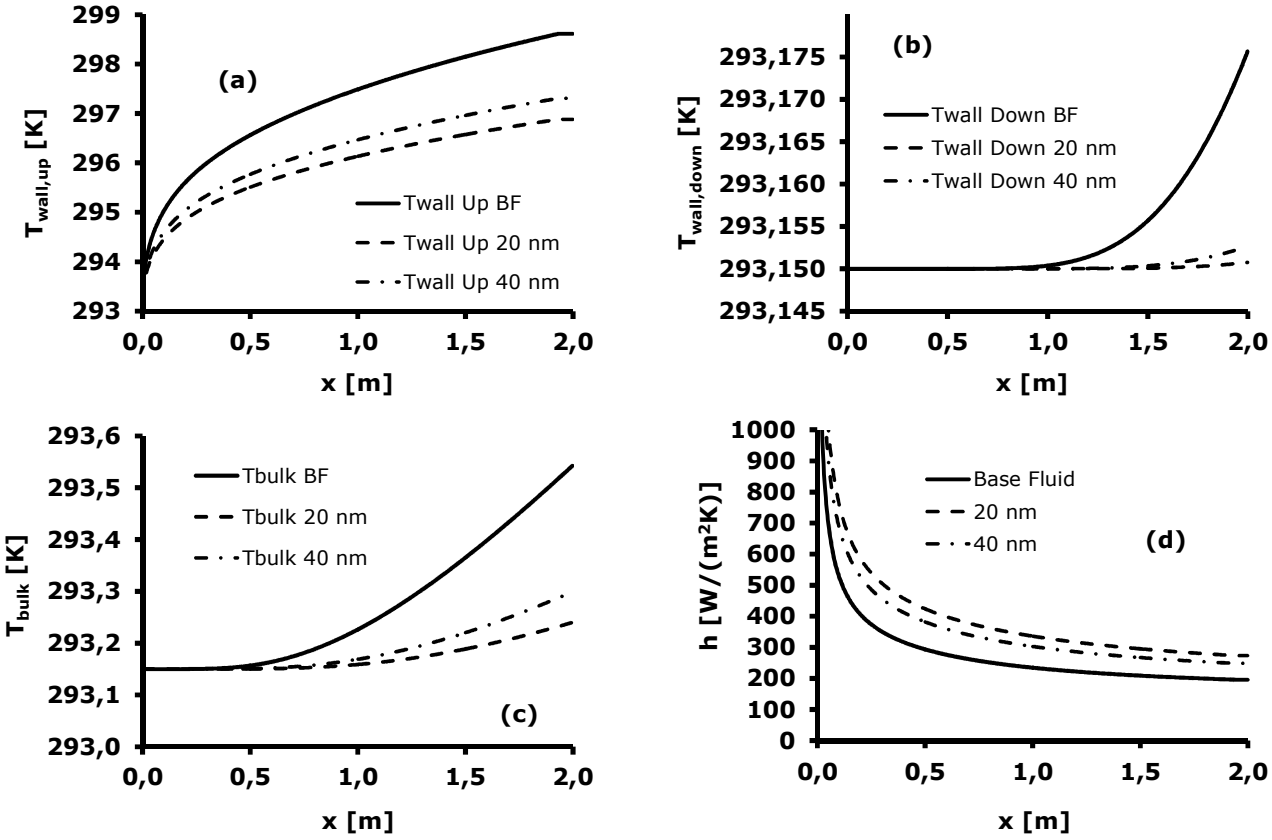

Figure 4. Impact of particles dimensions on the temperature and heat transfer coefficient profiles: (a) temperature on the top wall; (b) temperature on the bottom wall; (c) bulk temperature; (d) local heat transfer coefficient

Fig. 4(b) shows the temperature on the bottom surface. It can be observed that the increase of temperature is very limited, this is due to the higher Re number, which does not give the possibility to the warm fluid to have the necessary contact with the bottom wall. Similarly, also the bulk temperature, Fig. 4(c), experiences a limited increase of the temperature due to the fact that the fluid has a limited time to exchange heat with the top surface in order to heat itself due to the high Re.
Also in the case of bulk temperature and temperature on the bottom wall, the nanolfuid with smaller particles reach lower temperatures demonstrating improved cooling performances.
Finally, Fig. 4(d) highlights that at the decrease of the particles dimensions there an increase in the heat transfer coefficient, thus it can be said that the lower is the dimension of the particles and the higher are the thermal performances of the nanofluids. As already mentioned, the increased interaction surface between the base fluid and the smaller nanoparticles determines the increase of the thermal conductivity of the nanofluid.
In order to perform a global comparison of all the analysed cases, Figure 5 presents the average Nu and the pressure drop for Al_2O_3-water nanofluid with particle dimensions of 20 nm.

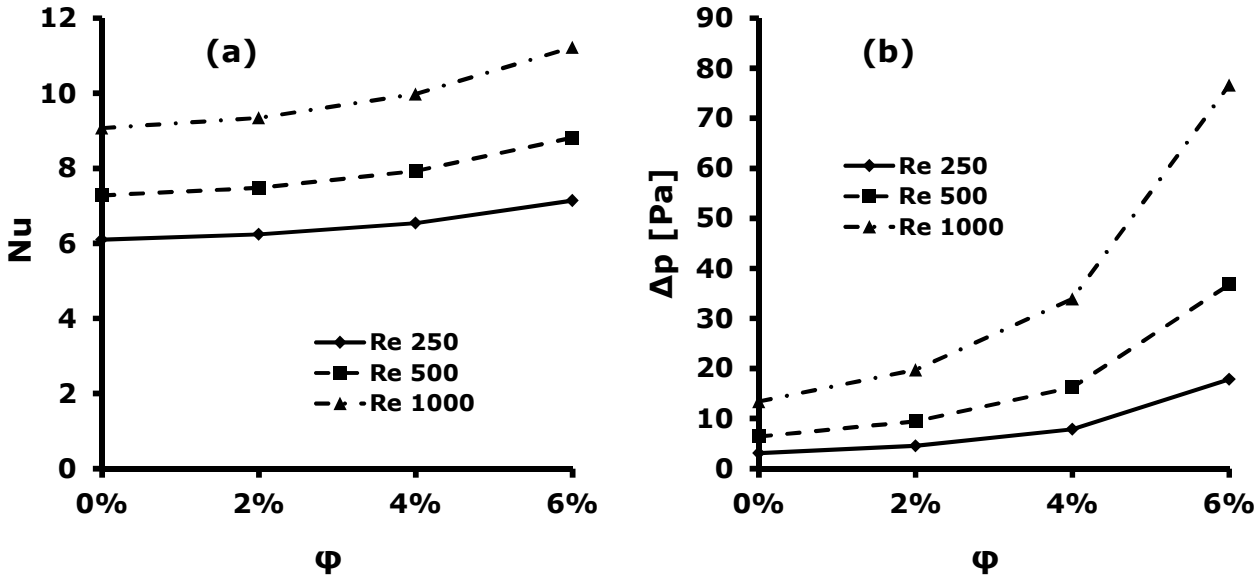

Figure 5. Performance of Al_2O_3- water nanofluid with particle dimensions of 20 nm: (a) average Nu; (b) pressure drop

It is detected from Fig. 5(a) that Nu number tends to increase when particles are added to the base fluid. In particular, for a concentration equal to 6%, an increase in Nu of the 17%, 21% and 24% is observed for Re equal to 250, 500 and 1000 respectively. Therefore, the heat transfer enhancement is relevant and this makes nanofluids attractive for a range of applications where cooling capabilities are prominent.

On the other hand, by analysing Fig. 5(b), it is detected that the pressure losses increase substantially as the particles concentration increases due to the increment of the viscosity. This means that the pumping power to use when nanofluids are considered is much higher with respect to the case of the base fluid (e.g. pure water).

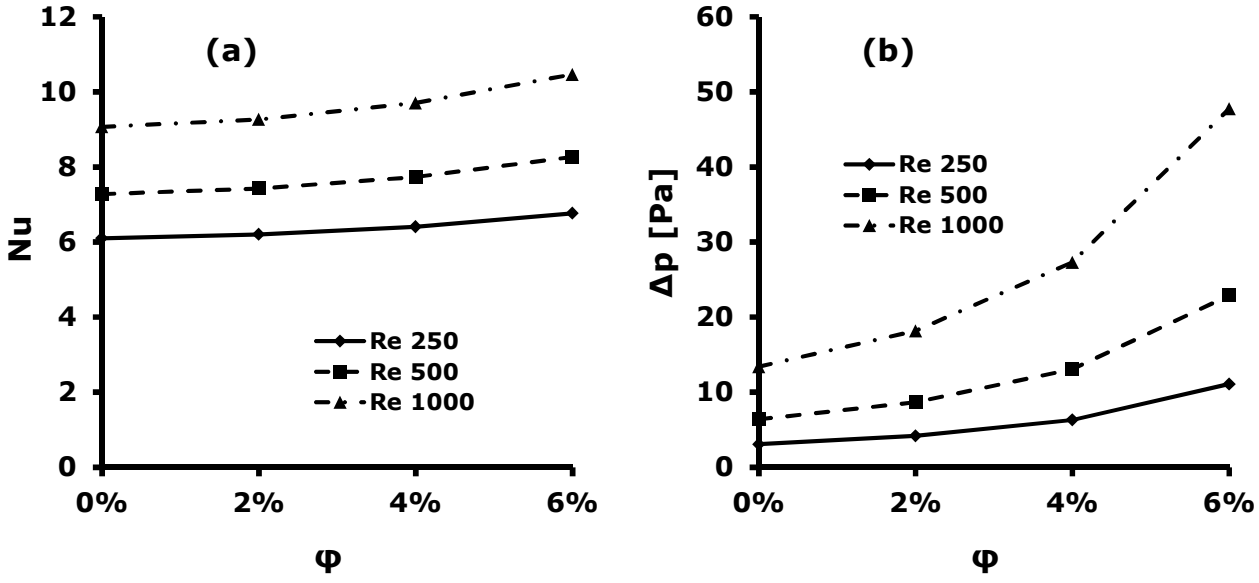

Figure 6. Performance of Al_2O_3- water nanofluid with particle dimensions of 40 nm: (a) average Nu; (b) pressure drop

Figure 6 shows the average Nu and the pressure drop when particles with a diameter of 40 nm are taken into account. By comparing Fig. 5 and Fig. 6, it can be detected that both the Nu and pressure drops values are lower when larger particles are considered. Therefore, the heat transfer is less effective, but pressure drops are lower, therefore less pumping power is required.

CONCLUSIONS

The present paper presented a numerical analysis of the laminar flow of Al_2O_3-water based nanofluid within a channel with a uniform heat flux of 1000 W/m^2 imposed on the top surface, while the bottom surface is adiabatic. A concentration ranging from 0% to 6% is considered, while particles of two dimensions, namely 20 and 40 nm, are taken into account. The analysis has been developed for Re equal to 250, 500 and 1000.
The analysis has demonstrated that nanofluids allow to obtain improved thermal performance. In particular, temperatures are lower with respect to the case of the base fluid and the heat transfer coefficients result to be higher. In particular, for a concentration equal to 6% an increase of the average Nu number of 17%, 21% and 24% is detected for Re equal to 250, 500 and 1000 respectively. On the other hand, a relevant increase of the pressure drop is also detected as the concentration increases and particles dimension decreases. In light of this, it is necessary to find the optimal trade-off between increasing of the heat transfer performances and the necessary pumping power.

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